Energy and Buildings 87 (2015) 226–232

Contents lists available at ScienceDirect

Energy and Buildings

journal homepage: www.elsevier.com/locate/enbuild

Experiments and thermal modeling on hybrid energy supply system of gas engine heat pumps and organic Rankine cycle

Huanwei Liu*, Qiushu Zhou, Haibo Zhao, Peifeng Wang

Department of Thermal Energy and Power Engineering, Yantai University, 30 Qingquan Road, Yantai 264005, PR China

ARTICLE INFO

Article history:
Received 4 May 2014
Received in revised form
17 November 2014
Accepted 19 November 2014
Available online 27 November 2014

Keywords:
Gas engine-driven heat pump
Waste heat recovery
Organic Rankine cycle
Thermal efficiency and exergy efficiency

ABSTRACT

This paper presents a hybrid energy supply system, which is composed of two subsystems (gas engine-driven heat pump system (GEHP) and organic Rankine cycle system (ORC)) and three major thermodynamic cycles (the vapor compression refrigeration cycle, the internal combustion gas engine cycle and ORC). In order to convert the low-grade gas engine waste heat into high-grade electricity, the ORC system is built up using R245fa, R152a and R123 as working fluids, and the ORC thermal model is also developed. Meanwhile, experiments of GHEPs in cooling mode are conducted, and several factors which influence the cooling performance are also discussed. The results indicate that the cooling capacity, gas engine energy consumption, gas engine waste heat increase with increasing of gas engine speed and decrease with decreasing of evaporator water inlet temperature. The waste heat recovered from gas engine is more than 55% of gas engine energy consumption. F6urthermore, R123 in ORC system yields the highest thermal and exergy efficiency of 11.84% and 54.24%, respectively. Although, thermal and exergy efficiency of R245fa is 11.42% and 52.25% lower than that of R123, its environmental performance exhibits favorable utilization for ORC using gas engine waste heat as low-grade heat source.

1. Introduction

The Gas Engine-driven Heat Pump (GEHP) is a vapor compression refrigeration type which includes compressor, condenser, expansion valve, evaporator and a gas engine to drive the compressor [1–7]. The GEHP is driven by a gas fuelled internal combustion engine instead of an electric motor. Although the efficiency of a gas engine is not very high (about 30–40%), the waste heat of fuel combustion can be recovered by approximately 50–60% through cylinder jacket and exhaust air heat exchanger. Therefore, the primary energy ratio (PER) of GEHP is higher than that of EHP. Compared with the electric-driven heat pump (EHP), the GEHP has two distinguished advantages: (1) the ability to recover the gas engine waste heat from cylinder jacket and exhaust gas, (2) and easy modulation of gas engine speed to accommodate the variation of cooling/heating loads. Therefore, many researchers have studied the gas engine-driven heat pumps from two main aspects of cooling or heating performance and thermodynamic simulations. Elgendy et al. [8–10] studied the heating/cooling performance of gas engine driven air to water heat pump with mixture refrigerant R410A, several factors on the performance of gas engine-driven heat pump system were performed. Lee et al. [11] researched the economic analysis of medium capacity space heating and cooling systems from three perspectives, such as primary energy, final consumer and social cost. Sanaye and Chahartaghi [12] has studied thermal modeling of gas engine-driven heat pumps including both the heat pump (consist of a compressor, condenser, expansion valve and evaporator) and engine systems. The dynamic modeling of GEHP system during startup in cooling mode and variation of evaporator and condenser temperature, shaft power consumed by compressor, engine fuel consumption and primary energy ratio of system were also researched [13]. A new system of air source energy independence driven by internal-combustion engine [14,15] has been presented, and a thermal model of a GEHP system working as water heater was investigated, furthermore, the heating capacity, coefficient of performance and primary energy ratio of the GEHP system were also studied by thermal simulation and experiment.

In the previous studies, the GEHP is focused on the performance research and thermal model simulation, the waste heat recovered from exhaust gas and cylinder jacket water is used for supplementary heating or defrosting [16,17]. Although, the supplementary heating or defrosting of evaporator improves the performance of GEHPs, the quality of low-grade gas engine waste heat could not be improved. On the other hand, because the ORC could transform

* Corresponding author. Tel.:+86 5356706013.
E-mail address: hwliu@ytu.edu.cn (H. Liu).

http://dx.doi.org/10.1016/j.enbuild.2014.11.046
0378-7788/

Nomenclature

B	natural gas consumption (kg/s)
COP	coefficient of performance
C_V	specific heat at constant volume (kJ/(kg °C))
C_p	specific heat (kJ/(kg °C))
h	specific enthalpy (kJ/kg)
LHV	lower heating value (kJ/m^3)
M	molecular weight (g/mol)
m	mass flow rate (kg/s)
m_v	volume flow rate (m^3/s)
N	speed (rpm)
P	pressure (MPa)
PER	primary energy ratio
Q	heat rate (kW)
Q_{use}	effective shaft power of gas engine (kW)
Q_c	cooling capacity (kW)
T	temperature (K)
t	temperature (°C)
u	parameters uncertainties (%)
W	power produced or consumed (kW)

Greek symbols

ϕ	ratio of effective to projected surface area of plate corrugations
η	efficiency
η_I	thermal efficiency
η_{II}	exergy efficiency

subscripts

a	air
all	all heat rate
cond	condenser
bp	normal boiling point
cj	cylinder jacket
com	compressor
crit	critical
evap	evaporator
eng	gas engine
exh	exhaust gas
g	mixture gas of air and natural gas
gas	natural gas
H	heat source
in	fluid inlet
l	heat lost of gas engine
L	heat sink
out	fluid outlet
p	refrigerant pump
t	turbine
r	refrigerant/working fluids
w	water

the low-grade thermal heat into high-grade electricity, the organic Rankine cycle is getting increasing attention with advantages of high efficiency, reliability and flexibility.

ORC as a promising energy conversion technology in the field of low-grade waste heat utilization has been studied by many researchers from different aspects. In recent years, some researchers [18–24] have paid more attention to employing organic Rankine cycle into utilize the diesel or gas engine waste heat. Saidur et al. [25] reviewed the latest developments and technologies on waste heat recovery of exhaust gas from internal combustion engines (ICE), including thermoelectric generators (TEG), organic Rankine cycle (ORC), six-stroke cycle IC engine and new developments on turbocharger technology. Tian et al. [26] proposed an organic Rankine cycle system used in the Internal Combustion Engine (ICE) exhaust heat recovery, and presented a simulation model based on an actual organic Rankine cycle bottoming system of a diesel engine. Furthermore, the ORC system was built to recover waste heat both from engine exhaust gas and jacket water using R245fa as working fluid. At the same time, many techniques were recently proposed to increase the internal combustion engines thermal efficiency and reduce CO_2 emissions. Tchanche et al. [27] presented existing applications of ORC and analyzed their maturity, including in car and engine field. Srinivasan et al. [28] examined the exhaust waste heat recovery potential of a dual fuel low temperature combustion engine using an ORC.

As mentioned previously, many researchers focused on the gas engine heat pump system performance, including the gas engine cylinder jacket waste heat and exhaust gas waste heat. But, the recovery waste heat is mainly used for the supplementary heating or defrosting of evaporator, the quality of low-grade gas engine waste heat could not be improved. Meanwhile, ORC as a promising energy conversion technology in the field of low-grade waste heat utilization has been studied by many researchers from different aspects. Therefore, in order to transfer the low-grade gas engine waste heat into high-grade electricity, the Lab gas engine heat pump system is further designed and modified. The paper first presented a hybrid energy supply system, including the gas engine heat pumps systems and ORC using gas engine waste heat as low-grade heat source. Second, cooling performance of gas engine heat pump system was studied, and the performance characteristics of GEHPs were also analyzed. Moreover, the variations between the waste heat recovery from gas engine and the water inlet temperature were obtained. Finally, thermodynamic model of ORC using gas engine waste heat are conducted, and the optimum value of each performance indicator for different fluids is compared respectively.

2. Experimental apparatus

Fig. 1 shows a schematic diagram of the compound experimental apparatus, which is composed of two subsystems, gas engine-driven heat pump system and ORC system. GEHPs is composed of a gas engine, an open type reciprocating compressor, a condenser, an electronic expansion valve, an evaporator, two heat exchangers (cylinder jacket heat exchanger and exhaust gas heat exchanger), measuring instruments, operational and safety control devices. Furthermore, the ORC system mainly includes vapor generator, turbine, generator, condenser, liquid receiver and refrigerant pump. All the pipes of the apparatus are thermally insulated. The water and ambient temperature were measured by a platinum resistance thermometer (with grade 'A' accuracy), the refrigerant and water temperatures at various locations were measured by Pre-calibrated PT100 sensors (with uncertainty ±0.5 °C), the water flow rates were measured by turbine flow meter (with grade '0.5′ accuracy), the low and the high pressures were measured by pressure transducers (with uncertainty ±0.08 bar). The testing points are shown in Fig. 1.

As shown in Fig. 1, the hybrid energy supply system includes three major thermodynamic cycles: the vapor compression refrigeration cycle, the internal combustion gas engine cycle and ORC.

GEHP can work as either a heat pump or a chiller. The heat pump system includes an open type compressor connected directly to the gas engine by a belt, which is different from the ordinary EHP (electric heat pump). Refrigeration circulation is comprised of an open compressor, an electronic expansion valve, an evaporator and a condenser followed by a receiver, a filter-driver and a sight glass. Refrigerant (R134a), absorbing heat in the evaporator, flows to the compressor through the reversing valve, the compressor increases

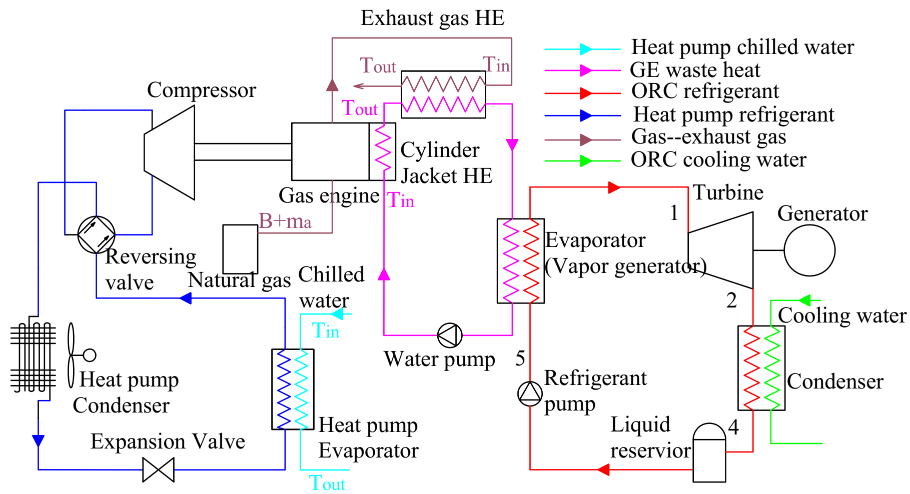

Fig. 1. Schematic diagram of the GEHPs and ORCs.

the pressure of the refrigerant and delivers superheated vapor to the condenser through an oil separator, then, passes through the electronic expansion valve. Finally, the refrigerant returns back to the evaporator.

In the GEHP, cylinder jacket heat exchanger and exhaust heat exchanger are adopted to recover the gas engine waste heat. The coolant discharged from the coolant pump is heated by the heat released from the cylinder jacket waste heat. At the same time, the coolant is further heated by the exhaust gas, then, the heated coolant passes through the vapor generator.

The gas engine waste heat could be used for vapor generator to vaporize the working fluids of ORC. The generated high pressure vapor in evaporator flows into the turbine and its enthalpy is converted into power. The low-pressure vapor is led to the condenser where it is liquefied by cooling water. Then the liquid is pumped into high pressure by refrigerant pump and then flows into the evaporator where it is vaporized by the gas engine waste heat. The *T*–*S* diagram is shown in Fig. 2.

As can be seen from the working processes of the ORC in Fig. 2, 1–2 is actual power produced in the expansion process, and 1–2s is the power produced for an isentropic expansion; 2–3–4 is the condensation process in condenser; the fluids is pressurized in the 4–5 process, and 4–5s is the fluid liquid pressurized for an isentropic compression; 5–6 is the liquid fluids heated into saturated liquid, and 6–1 is the process of the saturated liquid vaporized into superheated vapor by the gas engine waste heat.

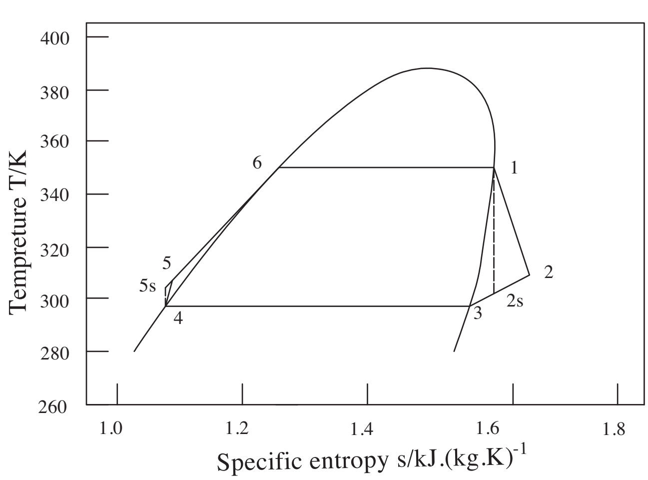

Fig. 2. *T*–*S* diagram of organic Rankine cycle.

3. GEHPs data reduction and ORCs thermal modeling

Cooling performance of gas engine heat pump system of R134a was experimental studied, and the performance characteristics (cooling capacity, gas engine energy input, coefficient of performance and primary energy ratio) of GEHPs were also analyzed. Furthermore, the variation between the waste heat recovery from gas engine and the water inlet temperature were obtained. Furthermore, based on the gas engine waste heat, the thermodynamic model of ORC was conducted, and the optimum value of performance indicator for different fluids was compared respectively.

3.1. GEHPs data analysis

Gas engine is the main component of the GEHP. The chemical energy of natural gas is converted into mechanical energy to drive the open compressor by the gas engine. Energy and mass balances are carried out for the main components of GEHP to compute their loads. The energy consumption of gas engine can be written as follows:

$$Q_{gas} = m_{v,gas}\mathrm{LHV} \quad (1)$$

The compressor input power W_{com} is tested by power instruments, the cooling capacity could be calculated by applying energy balance equation for the evaporator as follows:

$$Q_c = m_w C_{pw}(T_{in} - T_{out}) \quad (2)$$

Gas engine waste heat recovery system includes cylinder jacket heat exchanger and exhaust gas heat exchanger. The heat exchangers for recovering the waste heat can be described by the energy equilibrium of the gas engine.

$$Q_{gas} = Q_{use} + Q_{cj} + Q_{exh} + Q_{lose} \quad (3)$$

The heat recovered from the cooling water of the engine is:

$$Q_{cj} = m_{cj}C_{Pw}(T_{cj,out} - T_{cj,in}) \quad (4)$$

The exhaust gas heat is given by:

$$Q_{exh} = (B + m_a)(C_{Pg}T_{g,out} - C_{Pg}T_{g,in}) \quad (5)$$

COP and PER are main evaluation parameters of the characteristics of the GEHPs. Considering the gas engine waste heat, the COP and PER are calculated as follows.

$$COP = \frac{Q_c + Q_{cj} + Q_{exh}}{W_{com}} \tag{6}$$

$$PER = \frac{Q_c + Q_{cj} + Q_{exh}}{Q_{gas}} \tag{7}$$

3.2. ORCs thermal modeling

The refrigerants of R245fa, R152a and R123 are used for the working fluids of ORC, and some properties of the working fluids are shown in Table 1. In this present paper, the energy efficiency and exergy efficiency based on the first and second laws of thermodynamics are evaluated. Furthermore, heat and pressure loss in practical cycles, and the following hypotheses are made to simplify the analysis: the internal irreversibility and pressure drop are neglected, and each component is considered as a steady-state system.

The mathematical model of ORC is introduced by the following equations:

Turbine

$$W_t = m_r(h_1 - h_{2s})\eta_t \tag{8}$$

Where, efficiency of turbine is given as the ratio of the actual power produced in the expansion process to the produced for an isentropic expansion, as follows:

$$\eta_t = \frac{h_1 - h_2}{h_1 - h_{2s}} \tag{9}$$

Condenser

$$Q_{cond} = m_r(h_2 - h_4) \tag{10}$$

Refrigerant pump

$$W_p = m_r(h_5 - h_4) = \frac{m_r(h_{5s} - h_4)}{\eta_p} \tag{11}$$

Evaporator

$$Q_{evap} = m_r(h_1 - h_5) \tag{12}$$

Thermal efficiency and exergy efficiency are calculated in Eqs. (13) and (14).

$$\eta_I = \left(W_t - W_p\right)/Q_{evap} \tag{13}$$

$$\eta_{II} = \frac{\left(W_t - W_p\right)}{Q_{evap}\left(1 - \frac{T_L}{T_H}\right)} \tag{14}$$

4. Results and discussions

4.1. GEHPs experimental tests

Under the conditions of ambient temperature 30.1 °C, Fig. 3 represents the variation of cooling capacity with the evaporator water inlet temperature and gas engine speeds. As the evaporator water inlet temperature varies from 24 °C to 11.8 °C, the temperature difference between the chilled water and refrigerant will be reduced in the evaporator. Therefore, the evaporative temperature and pressure also decrease leading to a decrease in both refrigerant cooling effect and mass flow rate. Under the condition of evaporator water inlet temperature varying from 24 °C to 11.8 °C, the cooling capacity of GEHP decrease 31.1%, 28.6%, 29.4% and 25.6% at the gas engine speed 1200 rpm, 1400 rpm, 1600 rpm and 1800 rpm, respectively. In other aspects, the cooling capacity increases with an increase of gas engine speed from 1200 rpm to 1800 rpm. The reason is that the mass flow rate of refrigeration and fuel consumption increased with increasing of the engine speed. The cooling capacity increases from 12.6% to 18.8% under condition of the gas engine speed varies from 1200 rpm to 1800 rpm in the evaporator water inlet temperature 11.8 °C.

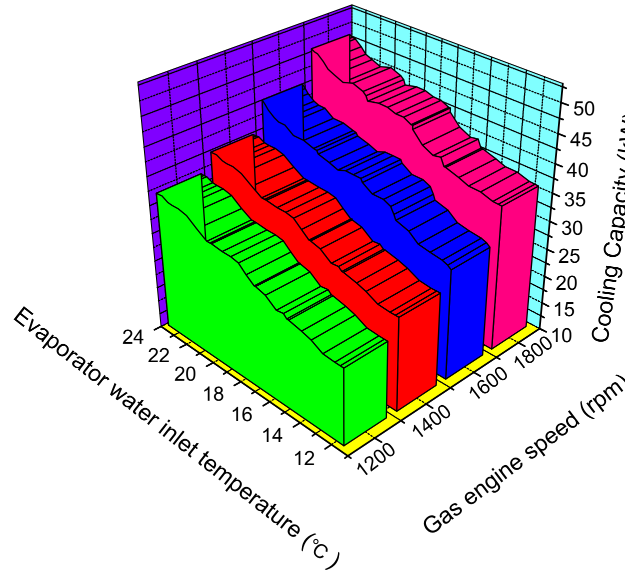

Fig. 3. Variation of cooling capacity versus evaporator water inlet temperature and gas engine speed.

The gas engine energy consumption and waste heat recovered from gas engine are shown in Figs. 4 and 5. The evaporative temperature and pressure also decrease leading to a decrease of both refrigerant cooling effect and mass flow rate, and the compressor is driven directly by gas engine. Therefore, considering the specific compressor power and mass flow rate, the compressor power, gas engine energy consumption vary in a small range under conditions of evaporator water inlet temperature from 24 °C to 11.8 °C. At the same time, the waste heat recovered from cylinder jacket heat exchanger and exhaust gas heat exchanger almost remain constant in different evaporator water inlet temperature. Analysis in depth of gas engine waste heat, the waste heat recovered from gas engine was more than 55% of gas engine energy consumption. In this paper, the low temperature energy was converted into electricity by ORC.

In addition, the variations of COP and PER with gas engine speed and evaporator water inlet temperature are shown in Figs. 6 and 7.

Table 1
Working fluids basic thermodynamic and environmental properties.

Refrigerants	M /g/mol	t_{bp} /°C	t_{crit} /°C	P_{crit} /Mpa	Safety data	ODP	GWP(100 yr)
R123	152.93	27.82	183.68	3.6	B1	0.02	77
R245fa	134.03	15.14	154.01	3.65	B1	0	1030
R152a	66.05	−24.02	113.26	4.5	A2	0	124

ODP: Ozone depletion potential.
GWP: Global warming potential.

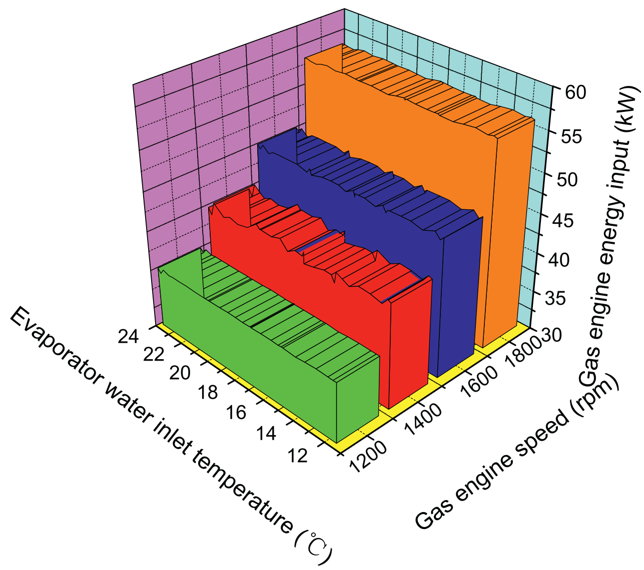

Fig. 4. Variation of gas engine energy input versus evaporator water inlet temperature and gas engine speed.

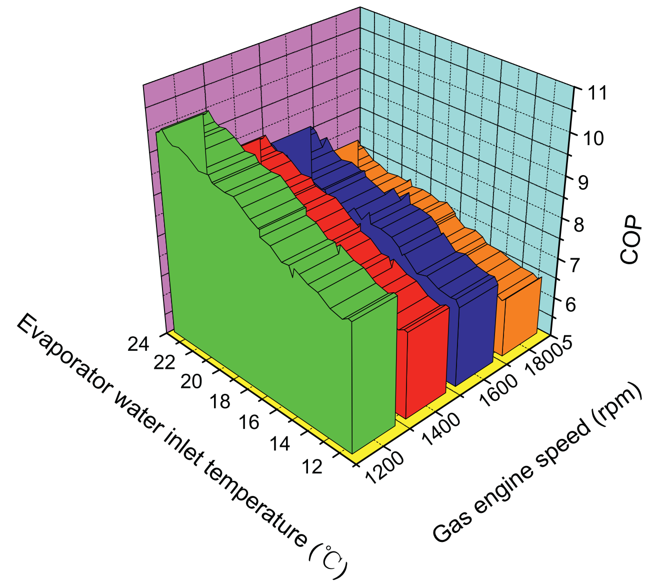

Fig. 6. Variation of COP versus evaporator water inlet temperature and gas engine speed.

As the evaporator water inlet temperature increases from 11.8 °C to 24 °C, a higher evaporator water inlet temperature yields a higher COP and PER, it is because that a large increase in cooling capacity compared to a small change in the compressor power and gas engine energy consumption. In other words, the quantity of the cooling load and waste heat increased lower than that of the compressor power and energy consumption increased. The variation tendency of COP and PER decreased with the increase in engine speed, the reason is that the rate of cooling capacity and waste heat is lower than the energy consumption of the engine. Under the condition of evaporator water inlet temperature 11.8 °C, although the cooling capacity and waste heat increase 49.1% and 44.1% from 1200 rpm to 1800 rpm respectively, the gas engine energy consumption and compressor power also increase 49.6% and 61.6% respectively.

4.2. ORCs thermal modeling analysis

As for the ORC system, the basic assumption as follows: turbine isentropic efficiency, mechanical efficiency and expansion ration is 88%, 98% and 5.5, respectively. Isentropic efficiency for the refrigerant pump is 80%. Condenser temperature is 299 K, and the thermodynamic properties of the working fluid R245fa, R152a and R123 are calculated by REFPROP NIST7.0. On the basis of the Eqs. (8)–(14), the ORC thermodynamic model is built by the software of matlab simulation. The thermal efficiency and exergy efficiency with evaporative temperature are shown in Figs. 8 and 9.

Fig. 8 represents the variation of thermal efficiency with the evaporative temperature. From the first law of thermodynamics, the high efficiency is obtained under the condition low condensing temperature and high evaporating temperature. As can be seen in Fig. 8, the thermal efficiency of R123 is higher than that of R245fa and R152a. The thermal efficiency of working fluid R245fa increases

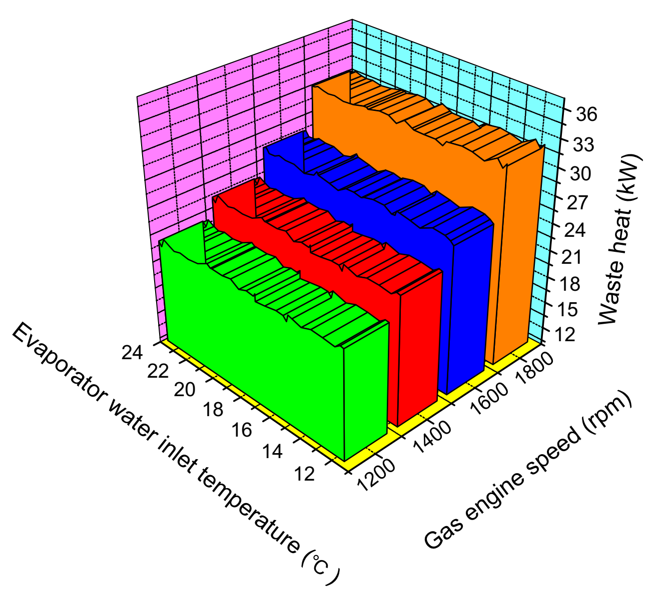

Fig. 5. Variation of waste heat versus evaporator water inlet temperature and gas engine speed.

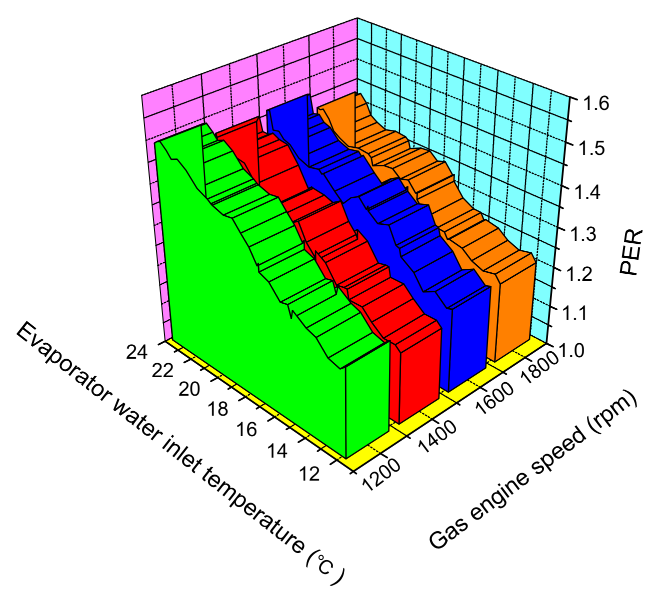

Fig. 7. Variation of PER versus evaporator water inlet temperature and gas engine speed.

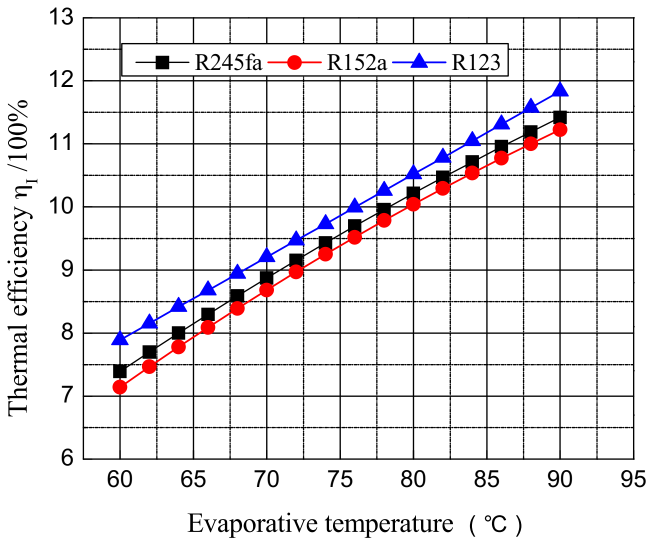

Fig. 8. Variation of system thermal efficiency with evaporative temperature.

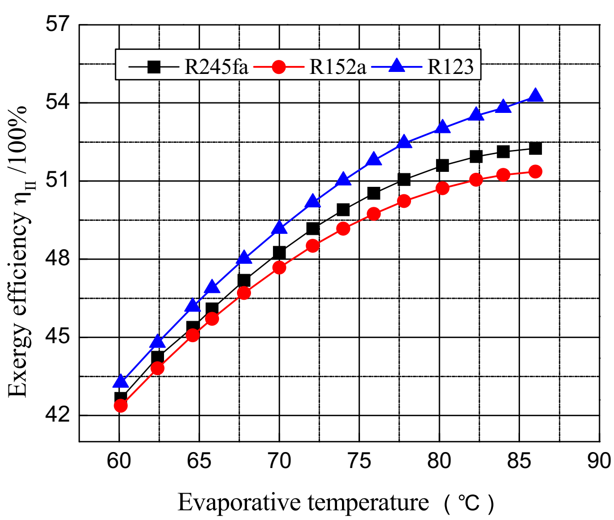

Fig. 9. Variation of system exergy efficiency with evaporative temperature.

from 7.39% to 11.42% with the evaporative temperature rising from 60 °C to 90 °C.

Fig. 9 shows the variation of exergy efficiency with the evaporative temperature, the exergy efficiency of R123 is also higher than that of R245fa and R152a. According to the definition of exergy efficiency, the exergy efficiency increases with the increasing of evaporative temperature, and exergy efficiency of R245fa varies in the range of 42.65–52.25%. Moreover, decreasing condensing temperature and increasing evaporating temperature also increase the exergy efficiency.

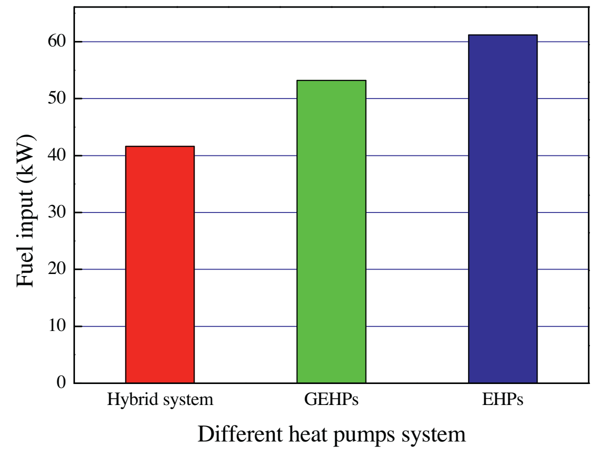

Fig. 11. The fuel input in different heat pump system.

4.3. Economic analysis of the hybrid energy supply system

Current practice is mainly to convert fuel to electricity at a power plant and reject the high-temperature waste heat to the environment. The energy conversion process is shown in Fig. 10. The electricity is transmitted to the heat pump motor, and the significant losses will occur from fuels to motor work of the heat pumps [3].

The hybrid energy supply system is composed of GEHPs and ORCs. In cooling mode, in the range of experimental data, the cooling capacity 40 kW and the electricity power 2.8 kW could be provided to the buildings. The fuel input between hybrid energy supply system, GEHPs and EHPs is shown in Fig. 11.

As is seen in Fig. 11, the fuel input of hybrid energy supply system is no more than 41.6 kW, which is less than that of GEHPs and EHPs. The main reason is that the waste heat recovered from gas engine is converted into the electricity by ORCs.

4.4. Uncertainty analysis

Uncertainties of cooling capacity, gas engine energy consumption, waste heat, COP and PER in this experiment could be obtained by equations as follows:

$$u(Q_c) = \sqrt{\left(\frac{\delta m_w}{m_w}\right)^2 + \left(\frac{\delta C_{pw}}{C_{pw}}\right)^2 + \left(\frac{\delta(T_{in} - T_{out})}{T_{in} - T_{out}}\right)^2} \tag{15}$$

$$u(Q_{gas}) = \sqrt{\left(\frac{\delta m_{gas}}{m_{gas}}\right)^2 + \left(\frac{\delta \mathrm{LHV}}{\mathrm{LHV}}\right)^2} \tag{16}$$

$$u(\mathrm{PER}) = \sqrt{(\sigma Q_c)^2 + \left(\sigma Q_{cj}\right)^2 + (\sigma Q_{exh})^2 + \left(\sigma Q_{gas}\right)^2} \tag{17}$$

The cooling capacity, gas engine energy consumption, waste heat, COP and PER were estimated with maximum uncertainty of 1.23%, 0.57%, 2.11%, 3.42% and 3.56%, respectively.

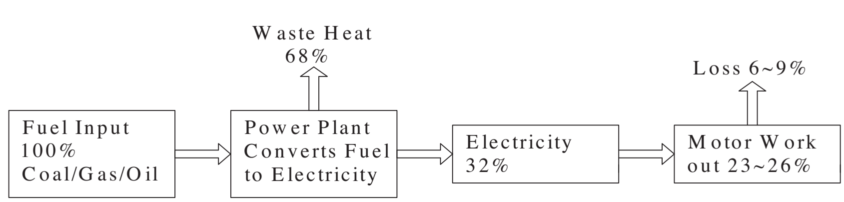

Fig. 10. Conversion process from fuels to motor work out.

5. Conclusions

A hybrid energy supply system, which is composed of two subsystems (gas engine-driven heat pump system (GEHP) and organic Rankine cycle system (ORC)) and three major thermodynamic cycles, has been put forward. The cooling performance of GEHPs and mathematic model of ORC were also described, and the main conclusions can be summarized as follows:

(1) The hybrid energy supply system including GEHPs and ORC is proposed, which could provide heating, cooling, hot water and electricity for buildings.
(2) Performance experiments and thermodynamic models of GEHPs are also investigated. The cooling capacity, COP and PRE were increased with an increase of evaporator water inlet temperature, however, the waste heat and gas engine energy consumption change in a small range with increase of evaporator water inlet temperature. Furthermore, the PER is more than 1.21 under the condition of evaporating water inlet temperature 11.8 °C, and the waste heat recovered from gas engine is more than 55% of gas engine energy consumption.
(3) Thermodynamic models of ORC with waste heat of GEHPs are established. R123 in ORC system with gas engine waste heat as low-grade heat source yields the highest thermal efficiency and exergy efficiency of 11.84% and 54.24, respectively. Although, thermal efficiency and exergy efficiency of R245fa is 11.42% and 52.25% lower than that of R123, its environmental performance exhibits favorable utilization for gas engine waste heat.

Acknowledgments

Supported by the Doctoral Fund of Yantai University of China (HX12B7), and by Research Projects in Colleges and University of Shandong Province (J12LB55). I am also grateful to the Ocean School of Yantai University.

References

[1] Aysegul Gungor, Zafer Ergbay, Arif Hepbasli, Exergoeconomic analyses of a gas engine driven heat pump drier and food drying process, Appl. Energy 88 (2011) 2677–2684.
[2] Aysegul Gungor, Zafer Ergbay, Arif Hepbasli, Exergetic analysis and evaluation of a new application of gas engine heat pumps (GEHPs) for food drying process, Appl. Energy 88 (2011) 882–891.
[3] A. Hepbaslia, Z. Erbayb, F. Icier, N. Colak, E. Hancioglu, A review of gas engine driven heat pumps (GEHPs) for residential and industrial applications, Renew. Sust. Energy Rev. 13 (2009) 85–99.
[4] Y.L. Li, X.S. Zhang, C. Liang, A novel parallel-type hybrid-power gas engine-driven heat pump system, Int. J. Refrig. 30 (2007) 1134–1142.
[5] Aysegul Gungor, Zafer Erbay, Arif Hepbasli, Huseyin Gunerhan, Splitting the exergy destruction into avoidable and unavoidable parts of a gas engine heat pump (GEHP) for food drying processes based on experimenal values, Energy Convers. Manag. 73 (2013) 309–316.
[6] Sepehr Sanaye, Mehdi Aghaei Meybodi, Mahmood Chahartaghi, Modeling and economic analysis of gas engine heat pumps for residential and commercial buildings in various climate regions of Iran, Energy Build. 42 (2010) 1129–1138.
[7] Zhi-Gao Sun, A combined heat and cold system driven by a gas industrail engine, Energy Convers. Manag. 48 (2007) 366–369.
[8] E. Elgendy, J. Schmit, Experimental study of gas engine driven air to water heat pump in cooling mode, Energy 35 (2010) 2461–2467.
[9] E. Elgendy, J. Schmit, A. Khalil, M. Fatouh, Performance of a gas engine heat pump (GEHP) using R410A for heating and cooling applications, Energy 35 (2010) 4941–4948.
[10] E. Elgendy, J. Schmit, A. Khalil, M. Fatouh, Performance of a gas engine heat pump (GEHP) using R410A for water supply, Energy 36 (2011) 2883–2889.
[11] Woo-Nam Lee, Hyeong-Jung Kim, Jong-Bae Park, Ki-Seon Cho, Jae Hyung Roh, Sung-Yong Son, Economic analysis of heating and cooling systems from the various perspectives: application to EHP and GHP in Korea, Renew. Sust. Energy Rev. 16 (2012) 4116–4125.
[12] Sepehr Sanaye, Mahmood Chahartaghi, Thermal modeling and operating tests for the gas engine-driven heat pump systems, Energy 35 (2010) 351–363.
[13] Sepehr Sanaye, Mahmood Chahartaghi, Hesam Asgari, Dynamic modeling of gas engine driven heat pump system in cooling mode, Energy 55 (2013) 195–208.
[14] Mingtao Wang, Zhao Yang, Xiangchao Su, Baihao Zhang, Xi Wu, Yongzheng Shi, Simulation and experimental research of engine rotary speed for gas engine heat pump based on expert control, Energy Build. 64 (2013) 95–102.
[15] Y.G. Chen, Z. Yang, X. Wu, M.T. Wang, H.W. Liu, Theoretical simulation and experimental research on the system of air source energy independence driven by internal-combustion engine, Energy Build. 43 (2011) 1351–1358.
[16] Zhao Yang, Wen-Bin Wang, Xi Wu, Thermal model and experiment of a gas-engine driven heat pump working as a water heater in winter, Energy Build. 58 (2013) 219–226.
[17] Zhenjun Xu, Zhao Yang, Saving energy in the heat-pump air conditioning system driven by gas engine, Energy Build. 41 (2009) 206–211.
[18] C.O. Katsanos, D.T. Hountalas, E.G. Pariotis, Thermldynamics analysis of a Rankine Cycle applied on a diesel truck engine using steam and organic medium, Energy Convers. Manag. 60 (2012) 68–76.
[19] Jacek Kalina, Integrated biomass gasification combined cycle distributed generation plant with reciprocating gas engine and ORC, Appl. Therm. Eng. 31 (2011) 2829–2840.
[20] Shekh Nisar Hossain, Saiful Bari, Waste heat recovery from the exhaust of a diesel generator using Rankine cycle, Energy Convers. Manag. 75 (2013) 141–151.
[21] E.H. Wang, H.G. Zhang, B.Y. Fan, M.G. Ouyang, Y. Zhao, Q.H. Mu, Study of working fluid selection of organic Rankine cycle (ORC) for engine waste heat recovery, Energy 36 (2011) 3406–3418.
[22] J.R. Serrano, V. Dolz, R. Novella, A. García, HD Diesel engine equipped with a bottoming Rankine cycle as a waste heat recovery system. Part 2: evaluation of alternative solutions, Appl. Therm. Eng. 36 (2012) 279–287.
[23] J. Wojeciech, Kostowski, Sergio Uson, Comparative evaluation of a natural gas expansion plant intergrated with an IC engine and an organic Rankine cycle, Energy Convers. Manag. 75 (2013) 509–516.
[24] Iacopo Vaja, Agostino Gambarotta, Internal combustion engine (ICE) bottoming with organic Rankine cycles (ORCs), Energy 35 (2010) 1084–1093.
[25] R. Saidur, M. Rezaei, W.K. Muzammil, M.H. Hassan, S. Paria, M. Hasanuzzaman, Technologies to recover exhaust heat from internal combustion engines, Renew. Sust. Energy Rev. 16 (2012) 5649–5659.
[26] Hua Tian, Gequn Shu, Haiqiao Wei, Xingyu Liang, Lina Liu, Fluids and parameters optimization for the organic Rankine cycles (ORCs) used in exhaust heat recovery of Internal Combustion Engine (ICE), Energy 47 (2012) 125–136.
[27] B.F. Tchanche, G.R. Lambrinos, A. Frangoudakis, G. Papadakis, Low-grade heat conversion into power using Organic Rankine Cycle—a review of various applications, Renew. Sust. Energy Rev. 15 (2011) 3963–3979.
[28] K.K. Srinivasan, P.J. Mago, S.R. Krishnan, Analysis of exhaust waste heat recovery from a dual fuel low temperature cimbustion engine using an Organic Rankine Cycle, Energy 25 (2010) 2387–2399.